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Optimisation of a Kalina cycle for a central receiver solar thermal power plant with direct steam generation

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Abstract:

Central receiver solar thermal power plants are regarded as one of the promising ways to generate electricity in near future. They offer the possibility of using high temperatures and pressures to achieve high efficiencies with standard power cycles. A direct steam generation approach can be used for such plants for improved performance. This approach can also be combined with using advanced power cycles like the Kalina cycle, which uses a zeotropic mixture of ammonia and water instead of pure water as the working fluid. This paper presents the optimisation of a particular Kalina cycle layout for a central receiver solar thermal power plant with direct steam generation. The variation in the cycle performance with respect to the turbine inlet ammonia mass fraction and pressure and a comparison of the initial investment with that of the basic Rankine cycle are also presented. Only high live steam conditions (temperature equal to 500 °C and pressure over 100 bar) were considered. The optimisation was performed using genetic algorithm. The results suggest that the optimum solutions for different turbine inlet ammonia mass fractions for the Kalina cycle occur at significantly different separator inlet conditions. Some of the Kalina cycle configurations can achieve similar or better cycle efficiency than the basic Rankine cycle but at a loss of the net electrical power output. The cost analysis suggests that some Kalina cycle configurations could have lower initial investment costs but only at higher turbine inlet pressure and inherently complex plant layouts as compared with the Rankine cycle.

Keywords: Solar thermal power plant, central receiver, Kalina cycle, optimisation

1. Introduction

Solar thermal power plants (STPPs) are considered to play an important role in the future energy mix [1]. An STPP consists of mainly three sections: the solar field, the storage system and the power cycle. The STPPs can be operated with either a multi-fluid configuration where the heat transfer fluid in the receiver is different from the power cycle working fluid, or with direct steam generation (DSG) configuration when using a Rankine cycle where pressurized steam is generated directly in the receiver and transported to the steam turbine. Application of DSG in STPPs presents the prospect of improving the overall plant efficiency, while simultaneously decreasing the cost of electricity generation [2]. In fact, the biggest central receiver STPP currently in operation is the Ivanpah plant in California, USA using DSG with water/steam as the working fluid [3]. The advantages of DSG include a higher live steam temperature and the use of one fluid as both the heat transfer fluid and the working fluid, possibly resulting in a simplified operation. The motivation behind the current study is that the exergy losses during a heat transfer process can be reduced by using a suitable multi-component working fluid which can evaporate and condense at a varying temperature, contrary to the constant evaporating and condensing temperature for a pure substance [4]. One such multi-component working fluid is the ammonia-water zeotropic mixture, as used in a Kalina cycle. Zhang et al. [5] presented a general review of using a Kalina cycle for different types of heat sources like the geothermal brine, waste heat recovery, etc. Modi and Haglind [6] presented a brief review of using the Kalina cycle in high live temperature and pressure power cycles and an analysis of using it in STPPs. Their results suggested that the Kalina cycle layout plays an important role in the assessment of the STPP performance and that using the Kalina cycle might be advantageous when operating from a two-tank molten-salt storage system.

There have been few studies presenting the methodology to solve or optimise the Kalina cycle for high temperature applications. Marston [7] presented the optimisation procedure for a Kalina cycle. A similar approach was used by Nag and Gupta [8] in performing an exergy analysis of the Kalina cycle. These studies assumed that the pinch point in a Kalina cycle condenser always occurs as the outlet for the ammonia-water mixture side. However, this isn't true for ammonia-water mixtures with high ammonia mass fraction. In fact, the pinch point can occur at any position in the heat exchanger depending on the ammonia mass fraction, especially if two-phase flow is involved. This depends on the convexity of the phase changing mixture temperature profile [9]. Thorin [10] presented the analysis of a bottoming Kalina cycle assuming a very small value of minimum pinch point temperature difference (PPTD), equal to 3 °C, for all the heat exchangers including boilers while using flue gases as the heat source.

This paper is aimed at defining the cycle parameters maximising the cycle efficiency of the Kalina cycle and comparing the Kalina cycle and a basic steam Rankine cycle in terms of initial investment cost. The results of optimisation of a particular Kalina cycle layout with two condensers and two recuperators to be used with a central receiver STPP with DSG are presented. A parametric study was also performed with respect to the pressure and the ammonia mass fraction at the turbine inlet. This study however doesn't include any evaluation of the power plant with storage or at part-load operating modes. Compared with the previous studies, more realistic assumptions regarding the heat exchanger pinch points are made and improved thermodynamic property calculation methods are employed.

2. Methodology

The Kalina cycle layout considered in this analysis is shown in Figure 1. The working solution ammonia-water mixture (stream 1) is expanded in the turbine. Energy is recovered from stream 2 to preheat the working solution in recuperator-1. In order to have a low condensation pressure in the condenser-1, a separator is used from which a rich ammonia vapour (stream 11) and a lean ammonia liquid (stream 12) are obtained. The lean liquid is mixed with the working solution (in mixer-1) and thus the ammonia mass fraction in condenser-1 is reduced. A throttle valve is used to bring the pressure of the lean liquid (stream 12) down to the pressure level of the working fluid (stream 4) before mixing in mixer-1. The rich vapour (stream 11) is mixed with the basic solution (stream 8) to again form the working fluid (stream 14) before going through the condenser-2 and then the pump-2 to increase the pressure equal to the turbine inlet pressure. After the recuperator-1, the working fluid (stream 17) is heated up to the turbine inlet temperature in the receiver/boiler.

The steam Rankine cycle considered in this study was a typical four-component cycle including a steam turbine, a wet-cooled condenser, a pump and a receiver/boiler.

The following assumptions were made for the cycle optimisation:

a. Both the cycles were modelled in steady state.
b. The turbine inlet temperature was fixed at 500 °C. The isentropic efficiencies of the turbine and the pumps were 85 % and 70 % respectively. The turbine mechanical efficiency and the generator efficiency were both 98 %. The plant was designed for a generator output of 20 MW. The minimum allowed vapour quality at the turbine outlet was 90 %. The condenser cooling water inlet and outlet temperatures were fixed at 20 °C and 30 °C respectively.
c. The recuperators and the condensers had a minimum PPTD of 8 °C and 4 °C respectively.
d. A pressure drop of 8 % [11] was considered for the DSG receiver. Pressure drops and heat losses in the other components were neglected for both the cycles.
e. For the Kalina cycle, the minimum separator inlet vapour quality was fixed at 5 %.

The two power cycles were modelled in MATLAB (2013b). The thermodynamic properties of both pure water for the Rankine cycle and the ammonia-water mixtures for the Kalina cycle were calculated using the REFPROP (9.1) interface for MATLAB [12]. The default property calculation method for the ammonia-water mixtures in REFPROP is using the Tillner-Roth and Friend formulation [13]. However, this formulation is highly unstable and fails to converge on several occasions, especially in the two-phase regions, near critical point and at higher ammonia mass fractions. Therefore, an alternative formulation called 'Ammonia (Lemmon)' [14] was tested and used. It was found to be more stable and with few convergence issues.

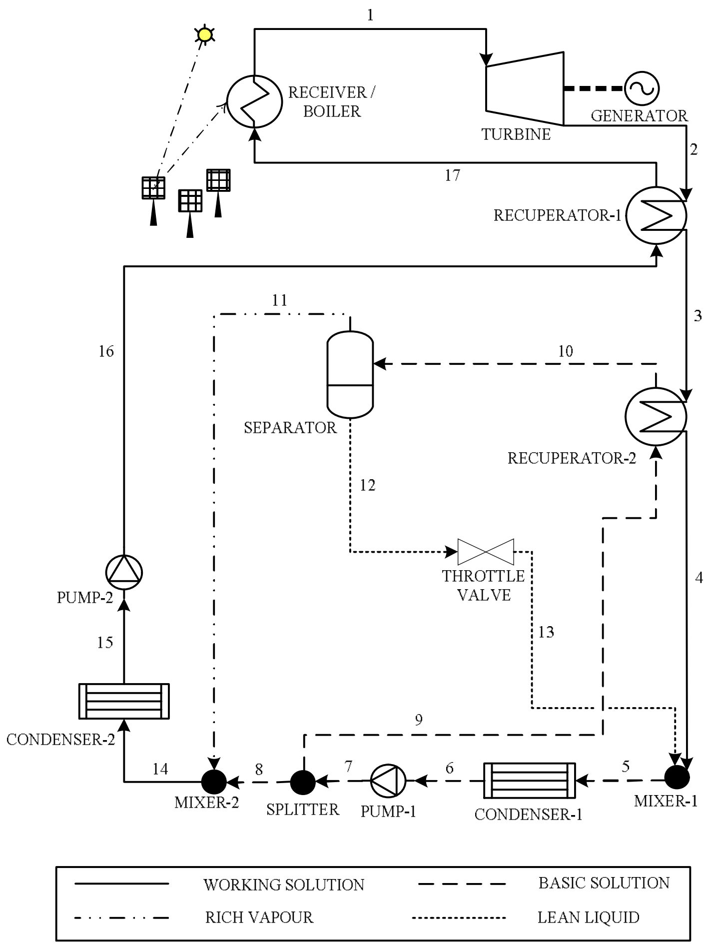

Figure 1: Kalina cycle layout

The Kalina cycle was optimised using the genetic algorithm from the Optimisation Toolbox of MATLAB. The objective of the optimisation was to maximise the cycle electrical efficiency (i.e. the ratio of the net electrical power output to the heat input in the receiver). The decision variables were the turbine outlet pressure, the separator inlet temperature and the separator inlet ammonia mass fraction. The optimisation was performed as follows:

a. The design parameters were provided as input to the genetic algorithm. These included the turbine inlet temperature, pressure and ammonia mass fraction, the generator power output, the efficiencies of the turbine, pumps and generator, etc. The lower and upper bounds for the decision variables were also provided as input.
b. The genetic algorithm then selected an initial population covering the entire search space and began the optimisation process, moving gradually towards the optimum solution.
c. The optimisation was performed in two steps: first with a wider range of the bounds to find out the region where the global maximum lies, then with a range close to the optimum solution. This was done to make sure that a global maximum is achieved and not a local maximum.
d. In case there was an error in the calculation of the thermodynamic properties by REFPROP, or the mass or energy balances were not satisfied with a residual below or equal to 0.001 %, the solution was rejected.
e. As a result of the optimisation, the thermal energy input to the receiver and the thermodynamic states at various points in the cycle were obtained.

For the Rankine cycle, the cycle was solved directly to find out the operating states which provide the minimum condenser pressure while satisfying the design constraints such as the turbine outlet vapour quality and the minimum PPTD.

A simplified cost analysis was then performed to compare the initial investment cost for the STPP when operating with either the Rankine cycle or the Kalina cycle. The cost functions used to calculate the purchase cost of the equipment are provided in Table 1.

Table 1: Cost functions for different components of the Kalina cycle and the Rankine cycle

Equipment	Cost function (cost in US$, work in kW and area in m^2)	Reference
Kalina cycle turbine	$4405 \cdot W_t^{0.7}$	[15]
Kalina cycle pumps	$1120 \cdot W_p^{0.8}$	[15]
Rankine cycle turbine	$3000 \cdot \left[1 + 5 \cdot \exp\left(\frac{T_1 - 866}{10.42}\right)\right] \cdot \left[1 + \left(\frac{1 - 0.953}{1 - \eta_t}\right)^3\right] \cdot W_t^{0.7}$	[16]
Rankine cycle pump	$623 \cdot W_p^{0.71} \cdot \left[1 + \left(\frac{1 - 0.8}{1 - \eta_p}\right)\right]$	[17]
Generator and electrical auxiliaries	$10 \times 10^6 \cdot \left(\frac{W_g}{160 \times 10^3}\right)^{0.7}$	[17]
Condensers and recuperators (assumed Shell and Tube type)	$32800 \cdot \left(\frac{A}{80}\right)^{0.68}$	[18]

The solar field for the two cycles were designed using DELSOL3 [19]. The heat required by the receiver as calculated from the power cycle optimisation was provided as an input to DELSOL3. Simulations were then run to determine the cost of the heliostat mirrors, the solar tower, the solar receiver, the required land area, the wiring and other fixed costs such as buildings, controls, etc. for the Rankine cycle and the different Kalina cycle configurations. The DELSOL3 simulations were run assuming the plant was located in Seville, Spain (coordinates 37.25° N, 5.54° W). The design direct normal irradiance was assumed to be 0.9 kW/m^2. The solar field was designed with a solar multiple of 1.3 and a flux limit on the receiver equal to 0.6 MW/m^2. The receiver was assumed as an external cylindrical type with a surrounding solar field.

In order to obtain the heat exchanger areas for the Kalina cycle, a constant overall heat transfer coefficient value (U-value) of 1 kW/m^2 was assumed for the recuperators and 1.1 kW/m^2 for the condensers [15]. For the Rankine cycle, a constant U-value of 1.25 kW/m^2 was assumed for the condenser [20]. The log mean temperature difference approach (assuming counter-flow) was then used to determine the heat exchanger area. Since the temperature profiles are not always linear in nature, the heat exchangers were divided into 50 control volumes each and the area of each control volume was determined. The total heat exchanger area was then the sum of the areas of all its control volumes.

All the obtained costs were updated to represent the equivalent cost for the year 2010 by using the respective Marshall & Swift economic indicators [21–24].

3. Results and discussion

The cycle efficiencies of the optimised Kalina cycle for different turbine inlet pressures and ammonia mass fractions are shown in Figure 2. The cycle efficiency of the Rankine cycle operating at a turbine inlet pressure of 100 bar was found to be 30.3 %. The efficiencies of the Rankine cycle for turbine inlet pressures over 100 bar were all less than that at 100 bar. This was mainly due to the design constraint of a minimum 90 % vapour quality at the turbine outlet which resulted in an increased condenser pressure for a feasible operation. Therefore, the case at 100 bar for the Rankine cycle is used for comparison with the different Kalina cycle configurations.

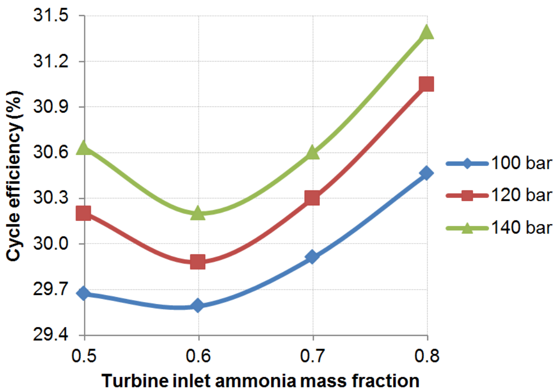

Figure 2: Optimum Kalina cycle efficiency at different turbine inlet pressures and ammonia mass fractions

From Figure 2, it may be observed that the cycle efficiency for the optimised cycle first decreases with the ammonia mass fraction and then increases. This is consistent with similar previous analyses by Modi and Haglind [6] and Marston [7]. The optimum values of the separator temperature and the separator inlet ammonia mass fraction vary significantly for different turbine inlet pressures and ammonia mass fractions. Therefore, if the same separator temperature is considered while varying the turbine inlet ammonia mass fraction, the trend will look different and result in an optimum value of turbine inlet ammonia mass fraction around 0.7 for the plant operation. Table 2 presents some operational and economic parameters of the Rankine cycle and the Kalina cycle for comparison.

Table 2: Comparison between the Rankine cycle and the Kalina cycle

Turbine inlet pressure (bar)	Turbine inlet ammonia mass fraction	Net electrical output (MWe)	Cooling water flow rate per MW of net electrical output (kg/s per MWe)	Total plant investment cost (million US$)
100	Rankine cycle	19.67	54.02	88.28
100	0.5	19.55	55.69	89.87
	0.6	19.49	55.89	89.87
	0.7	19.41	55.03	89.00
	0.8	19.33	53.59	87.55
120	0.5	19.47	54.28	88.28
	0.6	19.40	55.12	88.84
	0.7	19.31	53.98	87.75
	0.8	19.22	52.09	86.03
140	0.5	19.39	53.16	87.07
	0.6	19.31	54.26	87.79
	0.7	19.22	53.21	86.76
	0.8	19.12	51.25	84.94

As may be observed from Table 2, the total plant investment costs do not vary significantly with the different cycles. In fact, there are several Kalina cycle configurations which have smaller investment cost than that of the Rankine cycle. An interesting observation would be the net electrical power output from the plant: it varies significantly between the Rankine cycle and the various Kalina cycle configurations. So, even if there is a possibility of obtaining a Kalina cycle configuration with similar or better investment cost and cycle efficiency, it comes at a loss of the net electrical power output. This decrease in net electrical power output is a direct result of using two pumps in the Kalina cycle as compared with one in the Rankine cycle. Even in this situation, comparing the investment cost of the plant per MW of electrical output, there are still a few Kalina cycle configurations with better results, e.g., at a turbine inlet pressure and ammonia mass fraction of 120/0.8, 140/0.5 and 140/0.8.

Another important aspect of an STPP is the water requirement for the plant operation. Since it is likely that an STPP will be installed in arid regions due to the high availability of solar irradiance, it is important to consider the amount for water required by the STPP for its operation. The main two water requirements are for the cleaning of the mirrors and in the wet-cooled condensers. In this study, only the condenser cooling water requirement is compared.

It may be observed from Table 2 that the total condenser cooling water flow rate per MW of net electrical power output for both the Kalina cycle and Rankine cycle is comparable. In fact, there are some Kalina cycle configurations where the cooling water requirement is less than that of the Rankine cycle; this when there are two condensers in the Kalina cycle while only one in the Rankine cycle. This similar consumption is a result of the design constraint of having a minimum vapour quality of 90 % at the turbine outlet while fixing the cooling water inlet and outlet temperatures and the minimum turbine inlet pressure for this study to 100 bar. All these constraints produce a combined effect of significantly increasing the minimum PPTD in the Rankine cycle condenser to approximately 37 °C. In order to have a more reasonable value of the minimum PPTD in the Rankine cycle condenser, the turbine inlet pressure needs to be reduced to about 60 bar. In this case, the cooling water flow rate in the Rankine cycle condenser then becomes about 50 kg/s per MW of net electrical power output. This 50 kg/s cooling water flow rate for the Rankine cycle is still comparable to the Kalina cycle cooling water requirement. The low Kalina cycle cooling water requirement is due to the temperature glide of the ammonia-water mixture in the condenser which results in a better match between the temperature profiles of the working fluid and the cooling water.

Keeping all this in view, it still cannot be neglected that in this case we are comparing a complex Kalina cycle layout with a four-component Rankine cycle which might make the Kalina cycle an unattractive option for a central receiver STPP with DSG. It should however be noted that the current analysis does not include the effect of part-load performance of the two cycles and the effect of adding a storage system to the plant. In addition, only one cost function for the ammonia-water mixture turbine could be found in the literature (from Dorj [15]) which gives a turbine cost nearly equal to that of a steam turbine for the same capacity. However, as previously presented by Modi and Haglind [6], the volumetric flow rate through a Kalina cycle turbine in much smaller than that in a steam turbine for similar capacity, and hence it should be more compact and therefore cheaper. The Kalina cycle might perform better when using a two-tank molten-salt storage system as suggested in [6], so this might also change the scenario a little bit and will be considered in future work.

4. Conclusion

The results from the optimisation of a particular Kalina cycle layout with two condensers and two recuperators, to be used with a central receiver STPP with DSG, are presented and discussed. The Kalina cycle was optimised with maximising the cycle efficiency as the objective function and the cycle low pressure, the separator inlet temperature and the separator inlet ammonia mass fraction as the decision variables. A basic cost analysis provided the initial investment costs for both the Kalina cycle and the Rankine cycle. The cycle efficiency and the investment costs for the Kalina cycle were found for different configurations by varying the turbine inlet pressure and ammonia mass fraction.

The study focussed on operating the power cycle with live steam pressure above 100 bar. The results suggest that it is possible to find Kalina cycle configurations with the plant investment cost less, and the cycle efficiency better than the Rankine cycle. However, this comes at a loss of the net electrical output from the Kalina cycle. It is also possible to find Kalina cycle configurations with nearly the same cooling water flow rate requirement as the Rankine cycle. Though, without considering the part-load behaviour of the power plants and the possibility of addition of a storage system, it seems that the basic Rankine cycle can give almost the same performance as a complex layout Kalina cycle and hence make the Kalina cycle an unattractive option for central receiver STPPs with DSG. The unavailability of more accurate cost functions for the ammonia-water mixture components, especially the turbine, however makes the determination of the Kalina cycle investment costs a little skewed in favour of the Rankine cycle.

Nomenclature

A Heat transfer area, m^2

DSG	Direct steam generation
PPTD	Pinch point temperature difference, °C
STPP	Solar thermal power plant
T_1	Turbine inlet temperature, °C or K
W_g	Generator electrical output, kW
W_p	Pump work requirement, kW
W_t	Turbine work output, kW
η_p	Pump isentropic efficiency
η_t	Turbine isentropic efficiency

References

[1] Greenpeace International, SolarPACES, and ESTELA, 2009, Concentrating Solar Power: Global Outlook 2009, Amsterdam, The Netherlands.

[2] Zarza E., Valenzuela L., León J., Hennecke K., Eck M., Weyers H.-D., and Eickhoff M., 2004, "Direct steam generation in parabolic troughs: Final results and conclusions of the DISS project," Energy, **29**(5-6), pp. 635–644.

[3] Plotkin A. R., Toupin K. M., Gillum C. B., Rancatore R. J., Yang T., and Mier D., 2011, "Solar receiver steam generator design for the Ivanpah solar electric generating system," Proceedings of the ASME 2011 Power Conference, ASME, Denver, Colorado, USA, pp. 1–7.

[4] Chen H., Goswami D. Y., Rahman M. M., and Stefanakos E. K., 2011, "A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power," Energy, **36**(1), pp. 549–555.

[5] Zhang X., He M., and Zhang Y., 2012, "A review of research on the Kalina cycle," Renew. Sustain. Energy Rev., **16**(7), pp. 5309–5318.

[6] Modi A., and Haglind F., 2014, "Performance analysis of a Kalina cycle for a central receiver solar thermal power plant with direct steam generation," Appl. Therm. Eng., **65**(1-2), pp. 201–208.

[7] Marston C. H., 1990, "Parametric analysis of the Kalina cycle," J. Eng. Gas Turbines Power, **112**, pp. 107–116.

[8] Nag P. K., and Gupta A. V. S. S. K. S., 1998, "Exergy analysis of the Kalina cycle," Appl. Therm. Eng., **18**(6), pp. 427–439.

[9] Kim K. H., Han C. H., and Kim K., 2012, "Effects of ammonia concentration on the thermodynamic performances of ammonia–water based power cycles," Thermochim. Acta, **530**, pp. 7–16.

[10] Thorin E., Dejfors C., and Svedberg G., 1998, "Thermodynamic properties of ammonia–water mixtures for power cycles," Int. J. Thermophys., **19**(2), pp. 501–510.

[11] Koretz B., Afremov L., Chernin O., and Rosin C., 2011, "Molten salt thermal energy storage for direct steam tower systems," Proceedings of SolarPACES Concentrating Solar Power and Chemical Energy Systems, Granada, Spain.

[12] National Institute for Standards and Technology, 2013, "REFPROP MATLAB Interface" [Online]. Available: www.boulder.nist.gov/div838/theory/refprop/Frequently_asked_questions.htm#MatLabApplications. [Accessed: 30-Aug-2013].

[13] Tillner-Roth R., and Friend D. G., 1998, "A Helmholtz free energy formulation of the thermodynamic properties of the mixture {water+ammonia}," J. Phys. Chem. Ref. Data, **27**(1), pp. 63–96.

[14] Lemmon E., 2013, "Personal communication."

[15] Dorj P., 2005, "Thermoeconomic analysis of a new geothermal utilization CHP plant in Tsetserleg, Mongolia," The United Nations University.

[16] Lozano M.-A., Valero A., and Serra L., 1993, "Theory of exergetic cost and thermoeconomic optimization," Proc. Int. Symp. ENSEC, **93**.

[17] Pelster S., 1998, "Environomic modelling and optimization of advanced combined cycle cogeneration power plants including CO2 separation options," Ecole Polytechnique Federale de Lausanne.

[18] Smith R., 2005, Chemical Process - Design and Integration, John Wiley & Sons, Ltd., West Sussex.

[19] Kistler B. L., 1986, A user's manual for DELSOL3: A computer code for calculating the optical performance and optimal system design for solar thermal central receiver plants, Livermore, USA.

[20] Sinnott R. K., 2005, Coulson & Richardson's Chemical Engineering Volume 6, Elsevier Butterworth-Heinemann, Oxford, UK.

[21] Sommerfeld J. T., 1999, "Tracking the Marshall & Swift Equipment Cost Index," Cost Eng., **41**(4), pp. 29–31.

[22] Loh H. P., Lyons J., and White III C. W., 2002, Process equipment cost estimation - Final report.

[23] Chemical Engineering, 2009, "Economic Indicators," Chem. Eng., (February), p. 64.

[24] Chemical Engineering, 2012, "Economic Indicators," Chem. Eng., (January), p. 56.